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(54) COLD-GAS REFRIGERATOR

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The following amendments were allowed under Section 29 on 1 November 1977

Page 2, line 11, *delete* twice *insert* four times

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29 November 1977

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... materials such as steel wool, phosphorus-bronze gauze or lead balls are used, the heat capacity per unit volume depending upon the density of packing of said materials in the regenerator housing.

If the packing density is increased, however, the resistance to flow increases with the result that the pressure differential across the regenerator increases. This resistance to flow results in a restriction of the maximum permissible flow rate of the working medium and therefore of the speed of the motor which drives the displacer. Consequently, the motors generally used have to be comparatively large in size. A further drawback of the known regenerators is that they are expensive to manufacture on account of the materials and manufacturing methods used.

Another drawback, particularly for smaller refrigerators, is that a seal must be provided between the displacer and the cylinder wall to ensure that all the working medium flows through the regenerator. The seal adds to the cost of materials and manufacture and also creates frictional losses.

According to the invention there is provided a cold-gas refrigerator comprising a compression space of variable volume communicating with an expansion space of variable volume which has a lower average temperature than the compression space during the operation of the refrigerator, a displacer which is reciprocable in a cylinder to vary the volume of the expansion space, and a regenerator arranged in the communication between said spaces, through which communication a gaseous working medium flows to and fro between the compression space and the expansion space during the operation of the refrigerator, wherein the regenerator is formed by an annular gap between the displacer and the cylinder wall cooperating therewith, at least one of the two surfaces of the displacer and said cylinder wall which face each other and bound the gap being formed of a material having a good heat capacity, and the hydraulic diameter d_h of the annular gap, which diameter is approximately equal to twice the width of the gap, satisfying the formula:

$$\sqrt[3]{240\eta \, nD^2 \; l \, \frac{M^{\cdot}}{\rho RT}} \leqslant d_h \leqslant \frac{0{\cdot}4\eta \, l}{\rho n D^2}$$

[*Price*

wherein:

η= viscosity of the working medium,
ρ= specific density of the working medium,
M= molecular weight of the working medium,
R= gas content,
T= average temperature in the gap,
l= the length of the gap in the axial direction,
n= the number of cycles per second of the refrigerator,
D= diameter of the displacer.

The hydraulic diameter of a duct, the annular gap in the present invention, is defined as the ratio of twice the cross-sectional area of the flow through the duct divided by the wetted perimeter of the duct.

The advantage of the cold-gas refrigerator according to the invention resides in the simple and cheap regenerator construction and also in the fact that due to the absence of a seal between the displacer and the cylinder wall, frictional losses are considerably reduced.

The displacer may be constituted by a cylindrical body made of a synthetic resin, for example, Teflon (Registered Trade Mark), covered on its cylindrical surface with a layer of metal, for example, stainless steel.

It has been found that when the width of the annular gap is chosen so that the hydraulic diameter of the gap lies within the stated limits, a good heat transfer between the working medium and the gap walls is ensured, while furthermore the resistance to flow of the working medium in the gap is low. The latter means that a high flow rate is admissible to that the number of cycles per second of the refrigerator may be large, as a result of which the dimensions of the electric motor driving the machine will be small.

The space factor of the regenerator may be smaller than 10% and is preferably lower than 1%. The term "space factor" is to be understood to mean in this respect the ratio of the "free" volume in the regenerator, i.e., the volume of the space within the annular gap, to the overall regenerator volume, i.e., the free volume plus the volume of the material of which one or both of the two surfaces bounding the gap is or are formed.

In order that the invention may be readily carried into effect, embodiments thereof will now be described in greater detail, by way of example, with reference to the accompanying drawings, in which

Figure 1 is a sectional elevation of part of a known cold-gas refrigerator,

Figure 2 is a sectional elevation of part of a cold-gas refrigerator having a gap regenerator in accordance with the invention, and

Figure 3 is a sectional elevation of part of a two-stage cold-gas refrigerator having a gap regenerator in accordance with the invention.

Figure 1 shows a part of a known cold-gas refrigerator 10. It comprises a cylinder 10*a* of stainless steel having a small thickness to keep the thermal conductivity in the longitudinal direction small. A displacer 14 is reciprocable in the cylinder 10*a*. Below the displacer is a compression space 13 and above it is an expansion space 19. The displacer 14 comprises a base portion 15 through which extend axial ducts 16 which communicate with a regenerator 11*b* incorporated in the displacer portion 11. The regenerator 11*b* consists of a material having a large heat capacity, for example, steel wool, phosphorus-bronze gauze or lead balls. This type of regenerator has a very good regenerative power but also a rather large resistance to flow for the gaseous working medium. A seal 18 made of a synthetic resin is secured around the base portion 15 of the displacer to prevent leakage between the displacer and the cylinder wall, so that the working medium can pass only through the ducts 16 and regenerator 11*b* when flowing from the compression space 13 to the expansion space 19 and back. The expansion space 19 is bounded on its upper side by a freezer plate 20 for transferring the cold produced out of the space 19. The extra cold part of the refrigerator is surrounded by a casing 21 which encloses a heat-insulating vacuum space 22.

Figure 2 shows diagrammatically a cold-gas refrigerator which generally corresponds to that shown in Figure 1 but in which the regenerator 11*b* is omitted. Casing 24, cylinder 28, expansion space 30 and housing 23 bounding compression space 25 similar to the corresponding parts in Figure 1. The displacer 26 consists of a solid cylindrical body manufactured from synthetic resin of poor thermal con-

ductivity and surrounded by a thin stainless-steel skin 27. Between the cylinder 28 which is also manufactured from stainless steel, and the displacer 26 there is an annular gap 29. In its passage from the compression space 25 to the expansion space 30 the working medium flows through the gap 29, giving up thermal energy to the metal walls 28 and 27. During its flow in the reverse direction the working medium absorbs the thermal energy stored in the walls 28 and 27. In order to obtain a readily operating machine, the dimensions of the gap 29 must be such that the heat-transfer number $\Omega \geqslant 200$ and the pressure differential Δ_p across the gap as a result of flow loss $\leqslant 0.1p$, where p is the average pressure in the machine. It has been found that these two conditions are satisfied when the hydraulic diameter d_h of the gap 29, which diameter corresponds to approximately twice the gap width, satisfies the formula:

$$\sqrt[3]{240\eta\, nD^2\, l\, \frac{M}{\rho RT}} \leqslant d_h \leqslant \frac{0{\cdot}4\eta\, l}{\rho n D^2}$$

wherein:

η = viscosity of the working medium,
ρ = specific density of the medium,
M = molecular weight of the medium,
R = gas content,
T = average temperature of the gap,
l = length of the gap in the axial direction,
n = number of cycles per second of the refrigerator,
D = diameter of the displacer.

A practical example will now be described using helium as a working medium at an average pressure of 20 atmospheres and in which the average gap temperature is 200°K. This gives:

$R \approx 8000$
$M = 4$
$T = 200$
$\eta = 10^{-5}$
$\rho = 4.8$

The displacer diameter D is 10 mm and the number of revolutions per second n=10. The gap length l is assumed to be 200 mm. The maximum hydraulic diameter of the gap then is:

$$d_h \leqslant \frac{0.4\,\eta\, 1}{\rho n D^2} = \frac{0.4\times10^{-5}\times200\times10^{-3}}{4.8\times10\times100\times10^{-6}} = 16\times10^{-2}\ \text{mm}.$$

The well-known formula for the heat-transfer number Ω is:

$$\Omega = \frac{\alpha F}{C_p \dot{m}}$$

wherein

α = heat-transfer coefficient,
C_p = heat capacity
F = heat-transfer area,
$\dot{m}$ = gas mass flow rate.

This formula can be rewritten as follows:

$$\Omega=\frac{\dfrac{\alpha\, d_h}{\lambda}}{\dfrac{C_p\eta}{\lambda}\cdot\dfrac{\dot{m} d_h}{\eta\, S}}\cdot\frac{F}{S}$$

wherein

d_h=hydraulic diameter of gap,
λ=coefficient of heat conduction,
η=coefficient of viscosity,
S=cross-sectional area of the gap.

Now, if

$$Nu=\frac{\alpha\, d_h}{\lambda}$$

$$Pr=\frac{C_p\eta}{\lambda}$$

$$Re=\frac{\dot{m}\, d_h}{\eta\, S}$$

then

$$\Omega=\frac{Nu}{Re\,.\,Pr}\cdot\frac{F}{S}$$

For an annular gap and Pr=0.75, it holds that:

$$=\frac{7.5}{Re\times 0.75}\cdot\frac{2\,\pi\, D1}{\pi\, D\,.\,\frac{1}{2}\, d_h}$$

$$=\frac{40}{Re}\cdot\frac{1}{d_h}$$

But

$$Re=\frac{\dot{m}\, d_h}{\eta\, S}=\frac{\rho i\, d_h}{\eta\, S}=\frac{2\,\rho i}{\eta\,\pi\, D}$$

whence

$$\Omega=\frac{40\,\eta\,\pi\, D}{2\,\rho i}\cdot\frac{1}{d_h}$$

where *i* is the average volume flow on the warm side of the regenerator and equals $2\,n\,.\,Vo=2\,n\,\frac{\pi}{4}\,D^2 s$, Vo and *s* being the stroke volume and stroke respectively of the displacer. For

this type of refrigerator, $s \approx D$ and the average flow in the regenerator is approximately equal to half that at the warm end of the regenerator. Hence,

$$\Omega = \frac{40 \times 4\, \eta\, \pi\, D}{2\, \rho\, \pi\, nD^3} \cdot \frac{1}{d_h} = \frac{80\, \eta}{\rho\, nD^2} \cdot \frac{1}{d_h}$$

Substituting the numerical values given above, this yields:

$$\Omega = \frac{80 \times 10^{-5}}{4.8 \times 10 \times 100 \times 10^{-6}} \cdot \frac{200}{16 \times 10^{-2}}$$

$$= 208$$

which ensures a good heat transfer. For this pressure differential Δp occurring across the gap it holds that:

$$\Delta p = \frac{96}{Re} \cdot \frac{1}{d_h} \cdot \tfrac{1}{2}\, \rho \frac{i^2}{S^2} \approx \frac{12\, \eta\, nD^2\, 1}{d_h^3}$$

$$\text{So } \Delta p = \frac{12 \times 10^{-5} \times 10 \times 10^{-2} \times 200 \times 10^{-3}}{(16 \times 10^{-2})^3 \times 10^{-9}} = 8000 \text{ kg/m}^2$$

Since $p = 20$ atm. $= 2 \times 10^5$ kg/sq.m,

$$\frac{\Delta p}{p} = 0.04,$$

which is very much lower than the value of 0.1 which is still admissible for a readily operating machine.

Thus a cold-gas refrigerator is obtained which is of simple construction and which gives a good heat transfer in the regenerator and suffers very low flow losses.

Figure 3 shows a cold-gas refrigerator of the two-stage type, that is to say, having two expansion spaces 31 and 30, the volumes of which are varied by the displacer 26. This displacer again consists of a body made of a synthetic resin and surrounded by a thin metal skin 27. Between the displacer and the cylinder wall there is again in an annular gap 29 which serves as a regenerator in the same manner as the gap 29 in the refrigerator shown in Figure 2. The operation of the refrigerator shown in Figure 3 will be clear after the above explanation.

WHAT WE CLAIM IS:—

1. A cold-gas refrigerator comprising a compression space of variable volume communicating with an expansion space of variable volume which has a lower average temperature than the compression space during the operation of the refrigerator, a displacer which is reciprocable in a cylinder to vary the volume of the expansion space, and a regenerator arranged in the communication between said spaces, through which communication a gaseous working medium flows to and fro between the compression space and the expansion space during the operation of the refrigerator, wherein the regenerator is formed by an annular gap between the displacer and the cylinder wall cooperating therewith, at least one of the two surfaces of the displacer and said cylinder wall which face each other and bound the gap being formed of a material having a good heat capacity, and the hydraulic diameter d_h of the annular gap, which diameter is approximately equal to twice the width of the gap, satisfying the formula:

$$\sqrt[3]{240\, \eta\, nD^2\, l\, \frac{M^*}{\rho RT}} \leqslant d_h \leqslant \frac{0{\cdot}4\, \eta\, l}{\rho\, n\, D^2}.$$

wherein:

η = viscosity of the working medium,
ρ = specific density of the working medium,
M = molecular weight of the working medium,
R = gas constant,
T = average temperature in the gap,
l = the length of the gap in the axial direction,
n = the number of cycles per second of the refrigerator,
D = diameter of the displacer.

2. A cold-gas refrigerator as claimed in claim 1, wherein the space factor of the regenerator is smaller than 10%.

3. A cold-gas refrigerator constructed and arranged to operate substantially as herein described with reference to Figure 2 or Figure 3 of the accompanying drawings.

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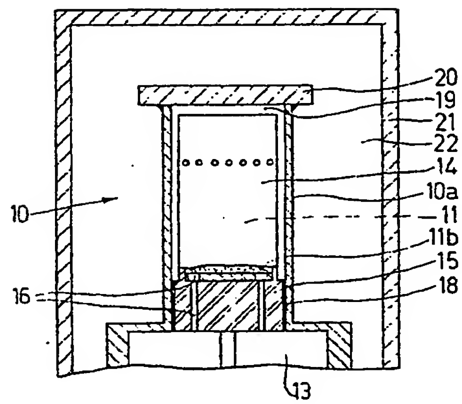

Fig.1

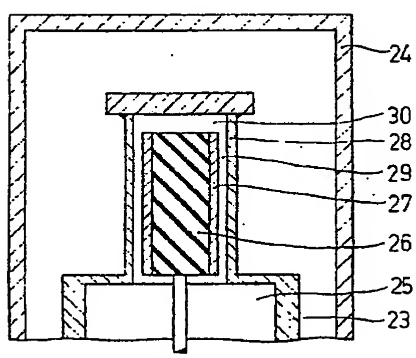

Fig.2

1335854 COMPLETE SPECIFICATION

2 SHEETS *This drawing is a reproduction of the Original on a reduced scale*

Sheet 2

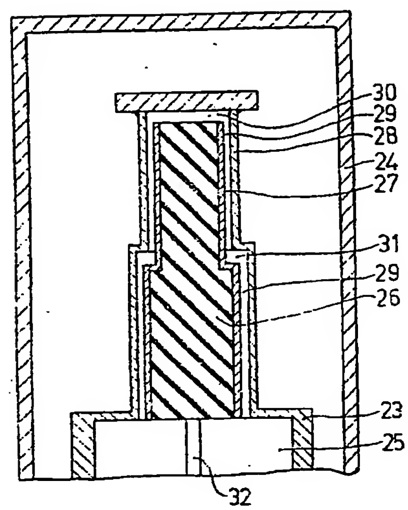

Fig.3